Slip Factor for Jet–Wake Flow in a Centrifugal Impeller

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Abstract

The centrifugal impeller outlet relative flow exhibits two regions, across blade to blade pitch. A region of low relative velocity towards the suction surface called wake flow and a region of high relative velocity towards the pressure surface called jet flow are known to exist. This leads to a high absolute velocity in the wake flow and a low absolute velocity in the jet flow with a large difference in absolute flow angles in these two regions. The slip factors for these two flows were experimentally evaluated through hot-wire anemometry measurement at different flow coefficients and they were compared with the values obtained from various correlations available in the literature. The measurements indicate that the slip factor for the jet and wake flows are different. The measured slip factor for the jet flow is lower than the values suggested by the correlations which is attributed due to viscous effects. The slip factor for the wake flow largely deviates from the value suggested by the jet-wake model. New slip correlations are being suggested for the jet and wake flow based on the measurements. The slip factor has been found to depend on flow coefficient and its value increases with decrease in flow coefficient.

Introduction

The relative flow angle at the outlet of a centrifugal impeller with respect to radial direction is greater than the blade outlet angle, due to the phenomenon of relative eddy which exists within the impeller channel induced by rotation. The effect of this is that the absolute whirl velocity at discharge is less than that calculated using blade angle itself. This difference in whirl velocity is called slip velocity, C_s of the impeller.

One-dimensional modelling impeller outlet flow is simple and useful for quick design optimisation of the compressor stage. Simple loss correlation to estimate the rotor efficiency plus a slip factor correlation to account for flow deviation[1,2] in the relative flow angle have been used to predict the impeller outlet flow conditions for a given requirement of mass flow rate and pressure ratio.

The real flow at impeller outlet was found to exhibit two regions across blade to blade pitch, viz., a low relative velocity region called wake towards the suction surface leading to high absolute velocity and angle and a high relative velocity region towards the pressure surface leading to low absolute velocity and angle. *Figure 1* shows the relative velocity variation across blade to blade pitch at impeller outlet. This behaviour of flow has been explained from the consideration of flow stability and equilibrium of pressure and Coriolis forces. The jet–wake models suggested by Dean[3] assumes that slip factors for the jet and wake flows are less than unity and unity respectively.

This paper describes the experimental evaluation of slip factors for the jet and wake flow at impeller outlet using hot-wire anemometry at different flow coefficients. The

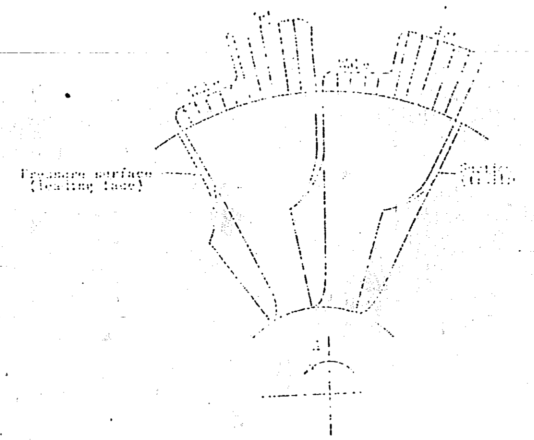

Fig. 1 Centrifugal impeller outlet flow in the radial exit plane

experimentally obtained slip factors have been compared with the values obtained from combination yaw probe and available correlations in the literature. New slip correlations have been suggested for the jet and wake flows based on the measurements.

Background

The slip factor, μ is defined by the relation

$$\mu = 1 - \frac{C_s}{U_2} \qquad \ldots\ldots\ 1$$

Wiesner[4], Stahler[5], Sakai *et al.*[6] have given correlations based on the blade geometry to evaluate the slip factor for centrifugal impellers. Wiesner[4] reviewed the various slip

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correlations available in the literature and checked with test data. He gave a correlation which fits very well with time averaged experimental results.

$$\mu = 1 - \frac{\sqrt{\cos \beta_{2b}}}{N_b^{0.7}} \qquad \ldots\ldots\ 2$$

The slip factor obtained from the above correlation is neither a jet nor a wake slip factor. Whitfield[7] considered flow through the jet to be similar to the potential flow and estimated the jet slip factor based on Stanitz[8] correlation, which uses relative circumferential jet width, θ_j

$$\mu_j = 1 - 0.315\theta_j \qquad \ldots\ldots\ 3$$

By considering the jet and wake flows at impeller outlet to mix out instantaneously just outside the impeller, Whitfield derived a relationship for the overall slip factor, μ_0

$$\mu_0 = \lambda \mu_W + (1 - \lambda)\, \mu_j \qquad \ldots\ldots\ 4$$

Whitfield and Wallace[9] showed that the slip factor obtained using Stanitz[8] correlation for the through flow jet is higher than the experimentally obtained value. They corrected the jet slip factor by an empirical parameter to get the overall slip factor ignoring any variation of slip in the wake region. This empirical parameter allowed for the reduced flow rate through the wake and non-uniform velocity distribution in the hub shroud plane. The correlation used is exactly similar to the one given by Wislicenus[10]. By assuming the velocity distribution at impeller outlet as sinusoidal, Wislicenus[10] derived an expression for the overall slip factor, μ_0

$$\mu_0 = 1 - \left\{\frac{\pi^2}{8}\right\}^2 (1 - \mu_j) \qquad \ldots\ldots\ 5$$

Balje[11] in his flow model for centrifugal impeller explained that the flow and kinetic energy stratification at rotor exit have a significant effect on the slip. The cross flow initiated by the flow stratification will induce relative peripheral components such that slip velocity is decreased at the suction side and increased at the pressure side indicating variation of slip in the circumferential direction.

Experimental Set-up

A centrifugal impeller of 525 mm diameter, 45.5 mm width with 23 blades backswept by 40 degrees with reference to radial direction was rotated at 5000 rpm by a D.C. motor. Thyristor control with feedback for the D.C. motor ensured maintenance of the speed to an accuracy of 0.1% (*Figure* 2). An electronic torquemeter coupled in between the gear box and the compressor was used to measure the speed and input power. A bell mouth in the inlet duct was used to ensure uniform flow to the impeller. A throttle plate at the exit of the volute casing was used to vary the mass flow rate through the impeller.

Instrumentation

The test facility was well instrumented for detailed flow measurements at impeller outlet. A hot-wire anemometer placed 8mm radially outwards of the impeller as shown in *Figure 3* was used in two angular positions to measure radial and tangential components of absolute velocity in a dynamic

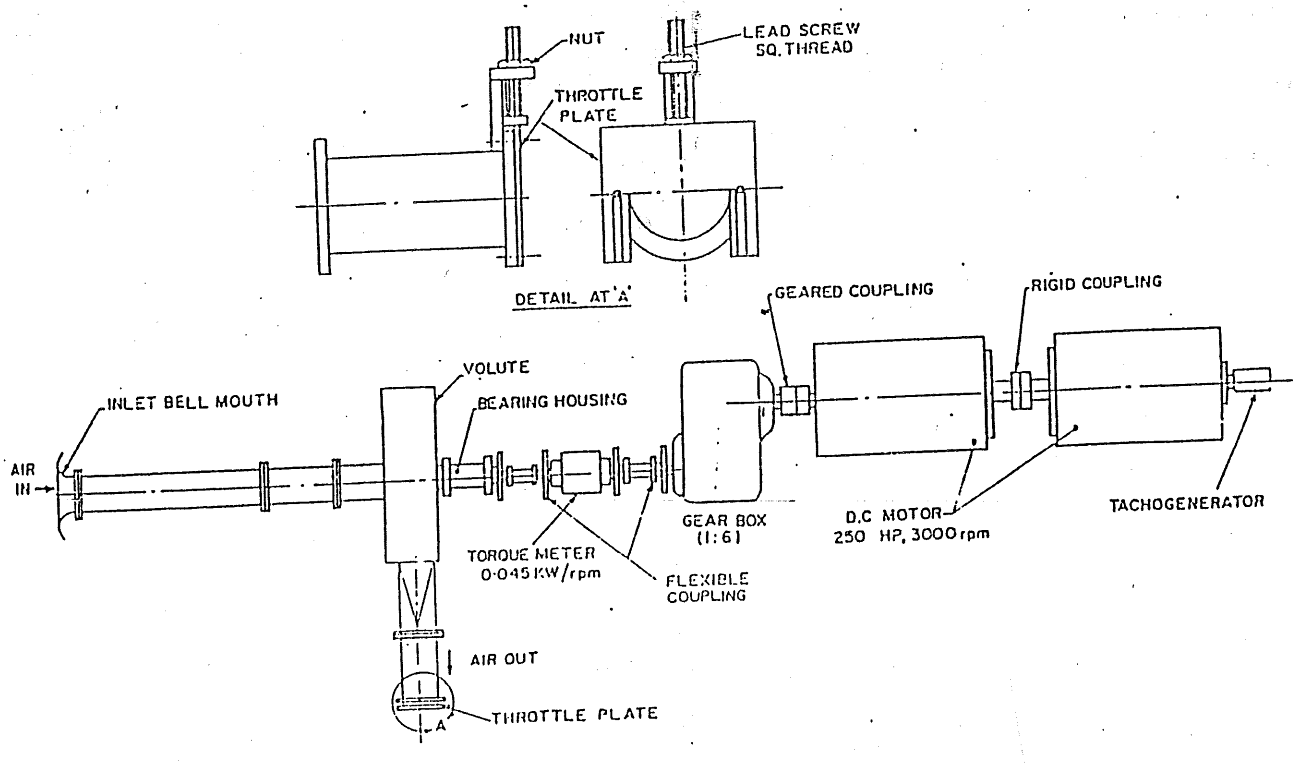

Fig. 2 Schematic layout of test facility

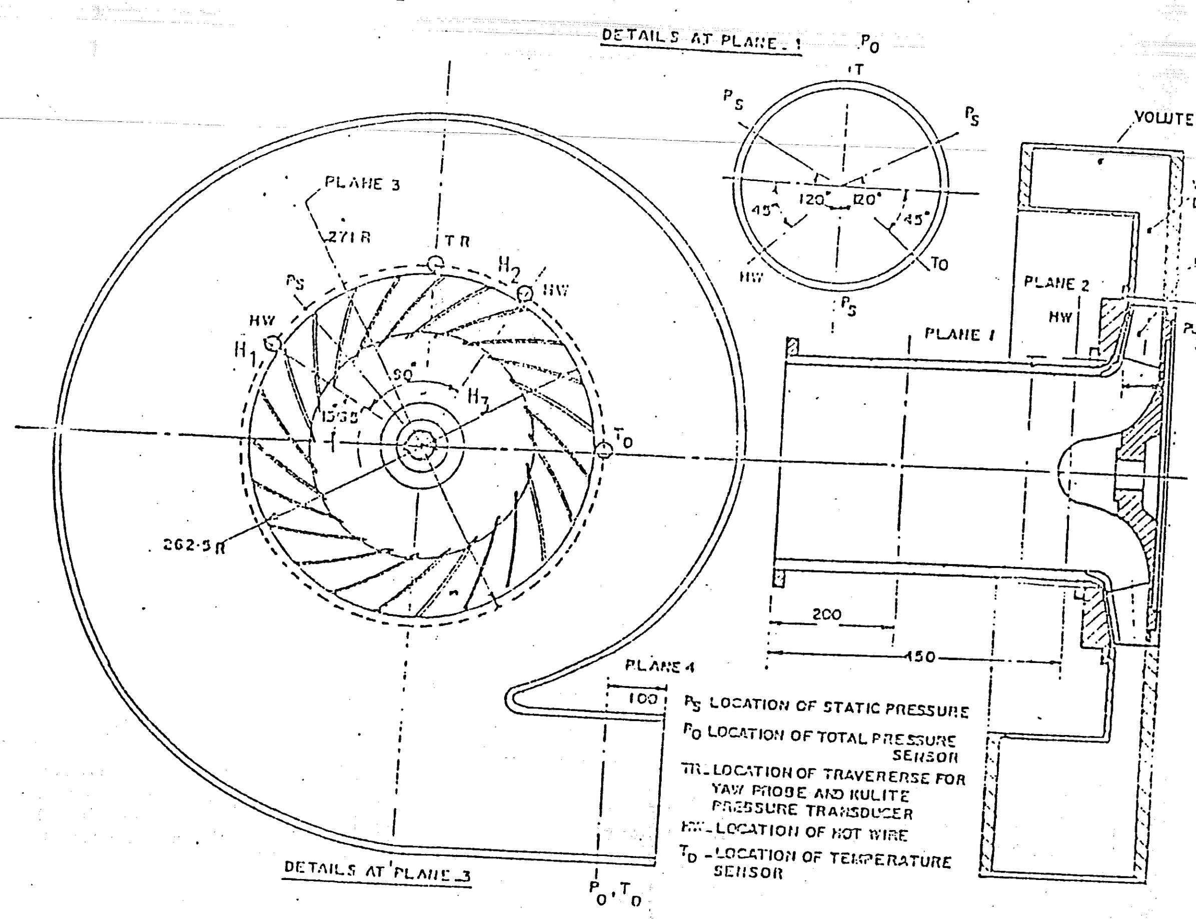

Fig. 3 Location of measurement planes

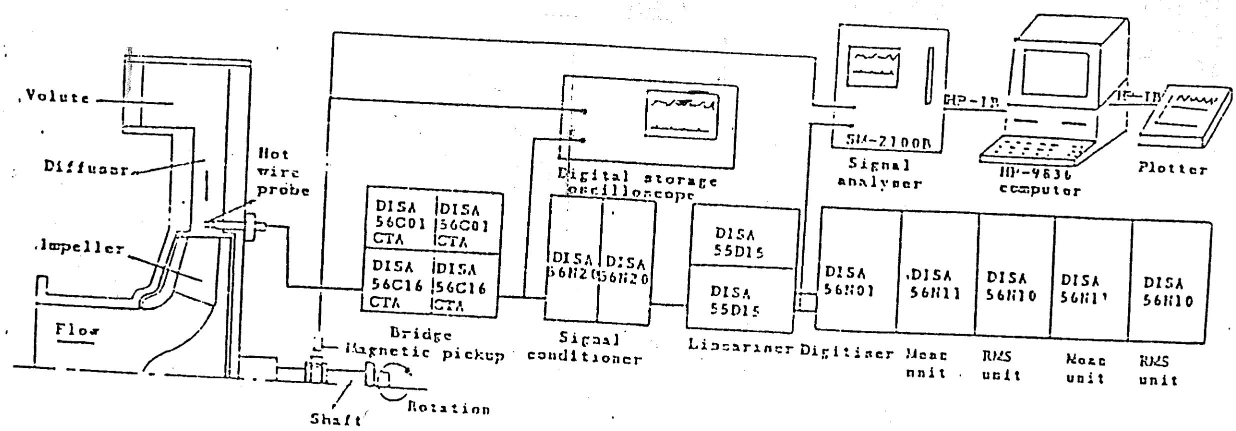

Fig. 4 Schematic layout of hot-wire anemometry system

mode across the blade to blade pitch at the mid channel between hub to shroud width of the impeller at outlet. A linearizer was used in conjunction with the hot-wire anemometer (*Figure 4*) circuit and calibration was carried out separately in a steady uniform flow before and after the experiments. The hot-wire traces were captured through a computer controlled dual beam signal analyser and recorded through its memory onto magnetic discs. An once per revolution spike generated from a magnetic pick-up and shaft projection was used to trigger the hot-wire trace and record the same for a duration of 50 ms corresponding to nearly revolutions of the impeller. Fifty such recordings one aft the other in the phase locked manner was obtained to get the ensemble average of the signal across 92 blade passage Additionally, a combination yaw probe was used to measu the time averaged velocity and flow angle which cou corroborate hot-wire measurements.

esults and Discussions

ot-wire traces of radial and tangential component of bsolute velocity for two blade passages near hub, mean and hroud section of the impeller width for three different flow oefficients are shown in *Figure 5* and *Figure 6*. The radial elocity distribution (*Figure 5*) indicate that at lower flow oefficients the total mass flow rate at impeller outlet is hared between the two regions with a lower value in the uction half of the blade passage and a higher value in the est of the passage. The tangential velocity distribution *Figure 6*) indicate that a good flow uniformity which exists at the mean width of blade passage at high flow coefficient, gets deteriorated as the flow coefficient is reduced and the loading is increased through higher incidence of flow. A region of high absolute velocity flowing at large angles is clearly seen to be growing by the side of the suction surface across blade pitch at impeller outlet. The time averaged radial and tangential components of absolute velocity across the axial width of the impeller are shown in *Figure 7* and *Figure 8*. It is observed from these figures that, axial distortion in the radial and tangential velocities are minimum at the optimum flow coefficient close to the design value of 0.10. The radial velocity is highly affected by the end wall boundary layers than the tangential velocity.

It can be shown from impeller outlet velocity vector diagram and using equation 1, the slip factor μ is given by

$$\mu = \frac{C_{\theta 2}}{U_2} + \frac{C_{r2}}{U_2} \tan \beta_{2b} \qquad \ldots\ldots\ 6$$

Assuming jet and wake flows as two separate regions and using the corresponding radial and tangential velocities at the mid channel, the slip factors for the jet flow is calculated at different flow coefficients. These are indicated by + symbols in *Figure 9*. Using the time averaged radial and tangential velocities obtained from yaw probe, the slip factors at different flow coefficients are calculated and are shown in *Figure 9* in asterisk. The slip factor variation with flow coefficient obtained from Stanitz correlation is also shown in *Figure 9*, by a thin line. It is observed from *Figure 9* that the jet slip factor obtained from instantaneous velocity measurements using hot–wire anemometry lie close to the values obtained from time averaged measurements using yaw probe. The slip factor obtained from stanitz correlation is very much higher than the values of slip factors obtained from measurements. This is because the correlation does not account for viscous effects, which effectively change the blade camber line shape due to boundary layer and also induce wall friction.

The slip correlation suggested by Stanitz for the jet flow was suitably modified to match with the experimentally obtained jet slip factors at different flow coefficients by accounting for the variation in blade geometry and jet width in the correlation. The modified slip correlation for the jet flow is given by

$$\mu_j = 1 - \frac{\sqrt{\cos \beta_{2b}}}{N_b^{0.7}} (1 + \theta_j) \qquad \ldots\ldots\ 7$$

Where θ_j is the circumferential relative jet width at impeller outlet. It depends on the flow coefficient. The variation of jet slip factor with flow coefficient obtained from the above correlation is shown in *Figure 9* by a thick line.

To calculate the slip factor in the wake flow, an average value of tangential velocity was used. From **Table 1** it is seen that the slip factor for the wake flow is greater than unity at lower flow coefficients and less than unity at highest flow coefficient, where the wake flow is almost absent.

Table 1 Estimated slip factors from measurements and from various slip correlations

Flow coefficient	0.128	0.094	0.078
Slip factor based on yaw probe measurements	0.80	0.82	0.88
Slip factor for jet flow	0.83	0.84	0.86
Slip factor for wake flow	0.91	1.16	1.18
Mass averaged slip factor	0.85	0.89	0.96
Slip factor based on Stanitz correlation	0.92	0.94	0.95
Slip factor based on Wislicenus correlation	0.88	0.90	0.92
Slip factor based on Wiesner correlation	0.91	0.91	0.91

From the values of jet and wake slip factors the mass averaged slip factor is calculated to get an overall slip factor. The mass averaged slip factors at different flow coefficient are shown in *Figure 10* by plus symbols (+). The overall slip factor is also calculated from Wislicenus[10] slip correlation which depend on jet slip factor. The variation of overall slip factor with flow coefficient obtained from Wiesner[4] and Wislicenus[10] correlation are shown in *Figure 10* with thin lines. Wiesner slip factor correlation depends only on the blade geometry and is independent of flow coefficient. Hence it is constant for a given impeller geometry. For the present impeller its value is around 0.91. It is observed from *Figure 10* that, at the point close to the design flow coefficient of 0.10 the values of slip factors obtained from Wiesner[4] and Wislicenus[10] correlations lie close to the value of overall slip factor obtained from measurement. But at off design points there is a large deviation between the values obtained from measurement and from correlations.

The overall slip correlation suggested by Wislicenus[10] is modified to match with the experimentally obtained mass averaged values of slip factor at different flow coefficients by taking jet width into account. The modified overall slip correlation is given by

$$\mu_0 = 1 - 1.26 \frac{\pi^2}{8} \theta_j^3 (1 - \mu_j) \qquad \ldots\ldots\ 8$$

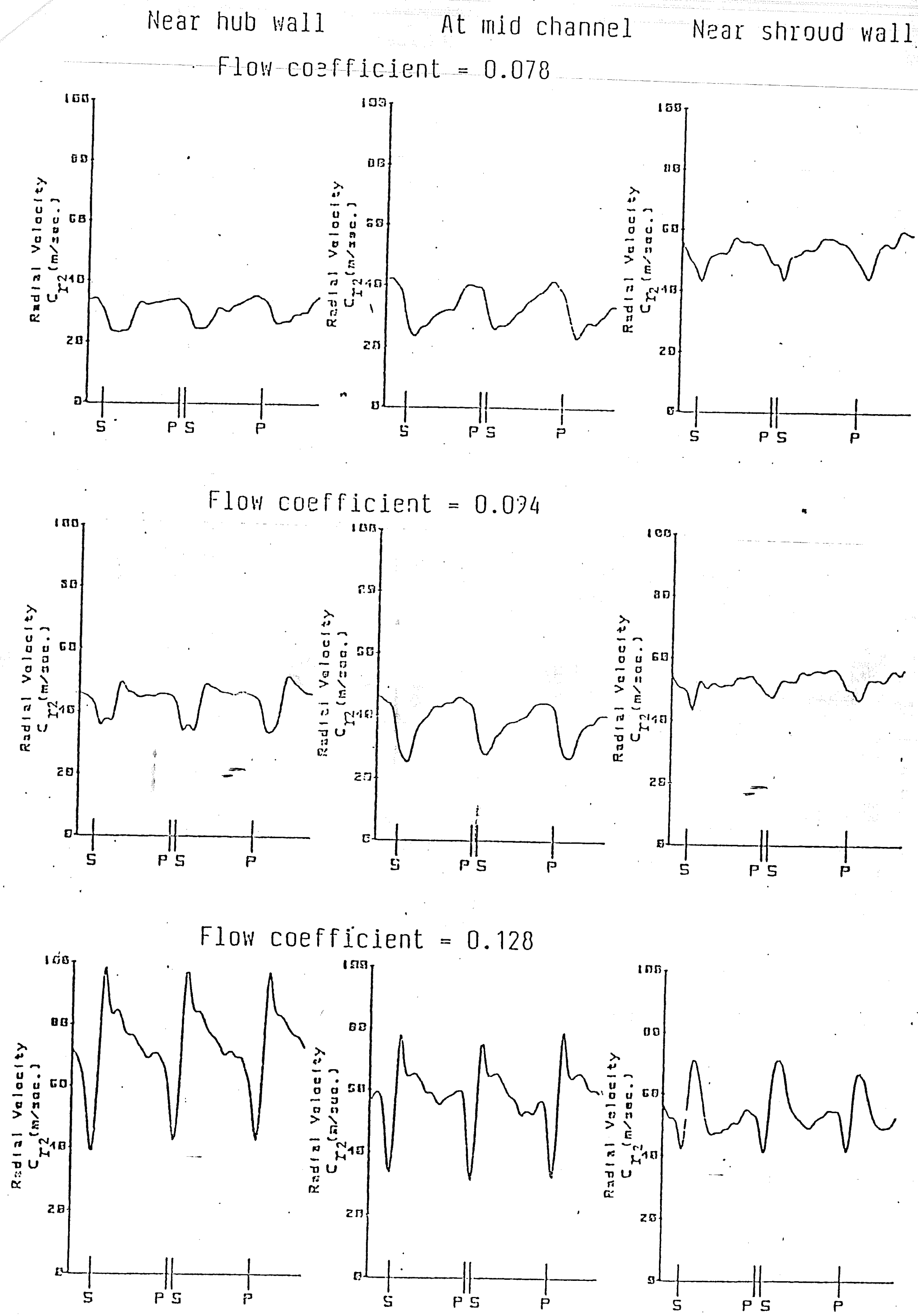

Fig. 5 Ensemble averaged instantaneous radial velocity variation at impeller outlet

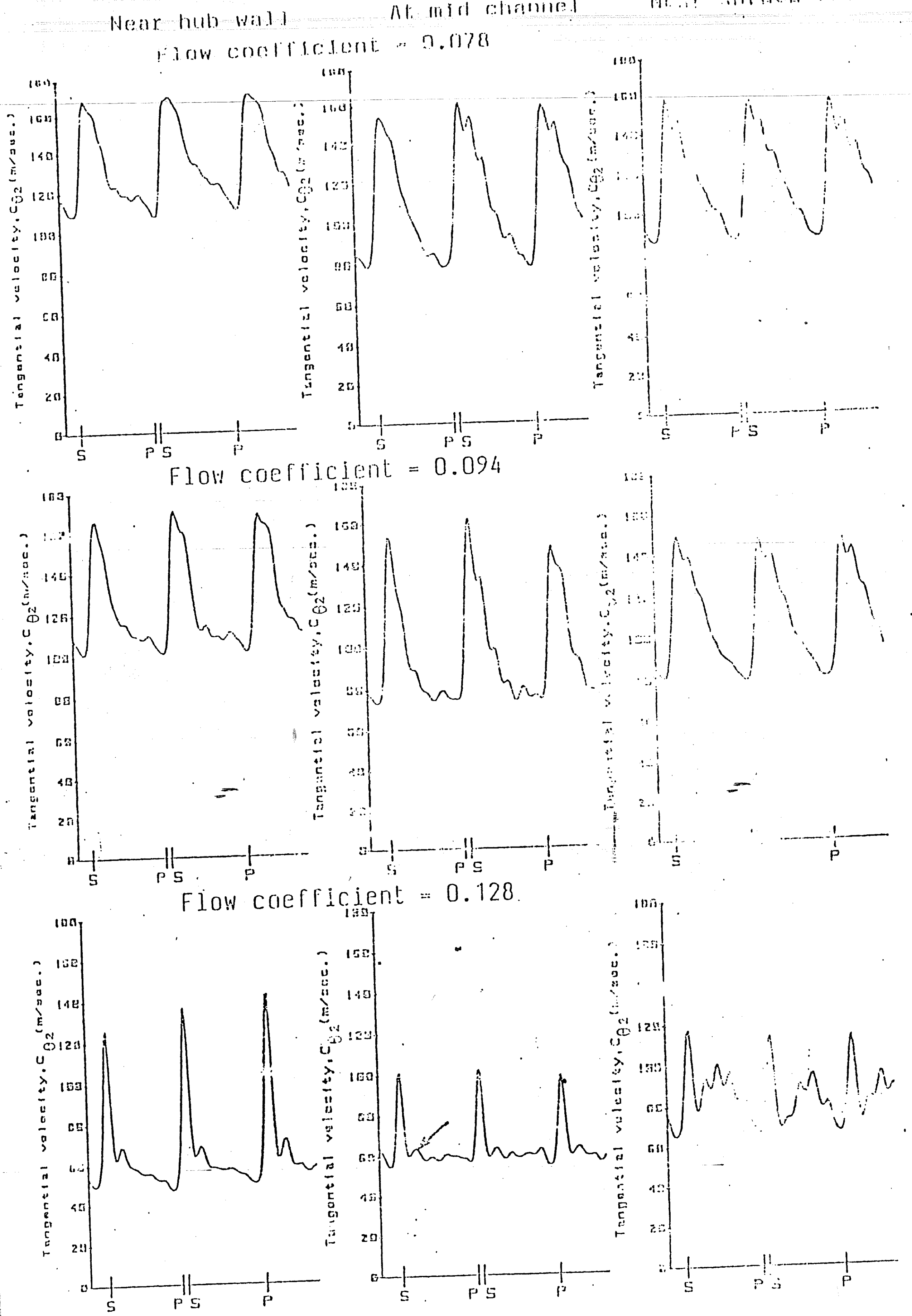

Fig. 6 Ensemble averaged instantaneous tangential velocity variation at impeller outlet

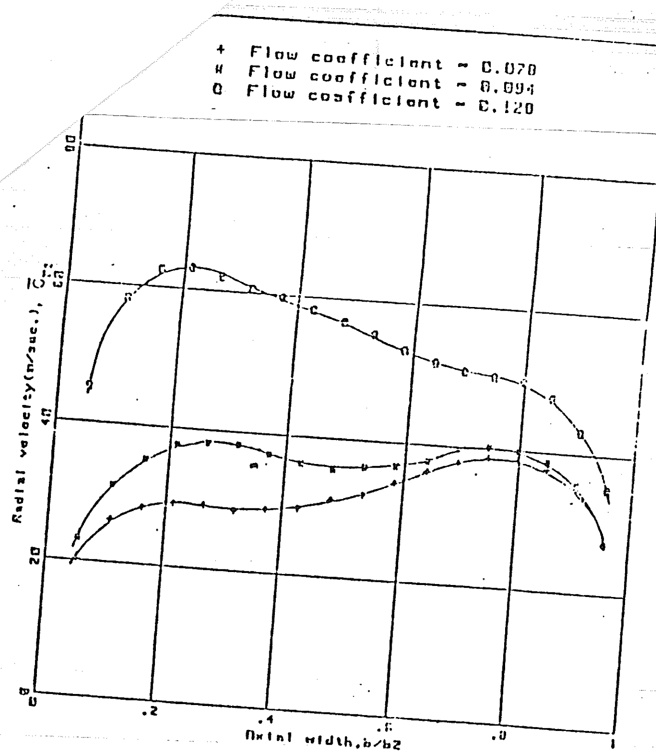

Fig. 7 Time averaged radial velocity distribution at impeller outlet

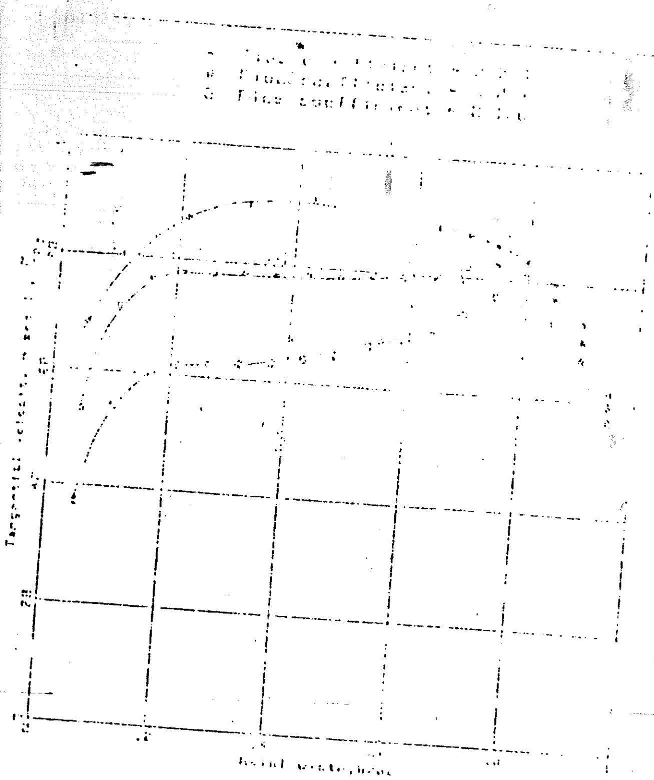

Fig. 8 Time averaged tangential velocity distribution at impeller outlet

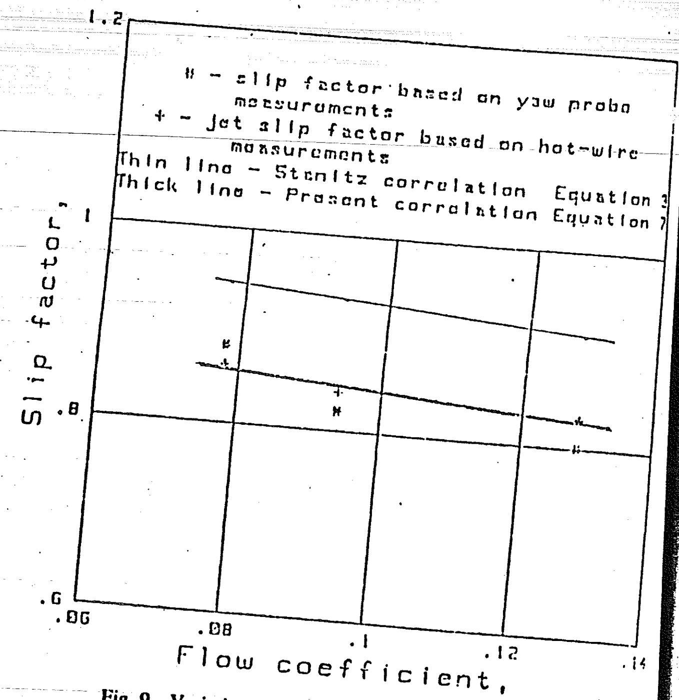

Fig. 9 Variation of jet slip factor with flow coefficient

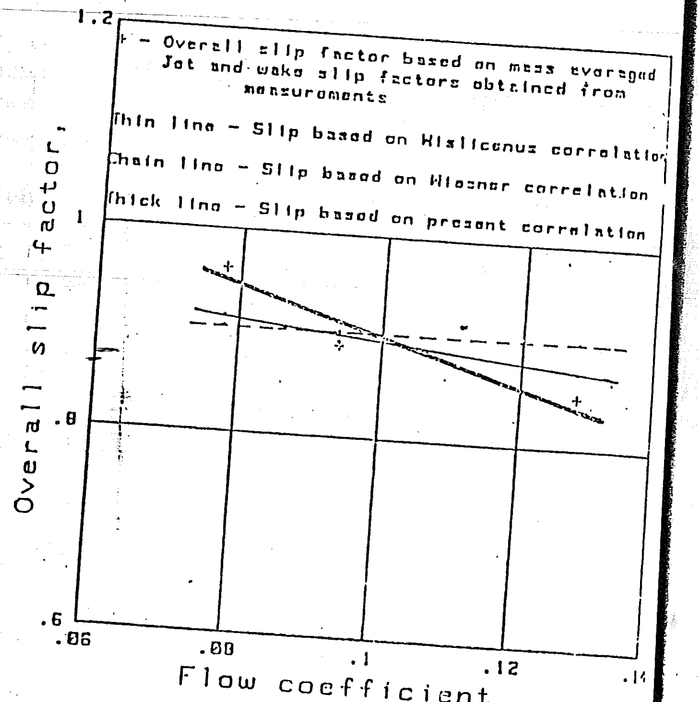

Fig. 10 Variation of overall slip factor with flow coefficient

The variation of overall slip factor with flow coefficient obtained from the above correlation is shown in *Figure 10* by a thick line.

The wake slip factor is now calculated from the values of jet and overall slip factors, which is given by

$$\mu_w = \frac{\mu_0 - (1 - \lambda)\ \mu_j}{\lambda} \qquad \ldots\ldots\ 9$$

Concl

The slip with de lower tl is high lower tl from St obtaine flow la width it flow ar specific region. measur high fr jet flow obtaine unity correla flow an relative obtain calcula presenc does n in the l

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Conclusions

The slip factor depends on the flow coefficient. It increases with decrease in flow coefficient. The jet slip factor is always lower than unity indicating that the outlet relative flow angle is higher than the blade angle and actual specific work is lower than the ideal specific work. The slip factor obtained from Stanitz correlation is very much higher than the values obtained from measurements. The slip factor for the wake flow largely depends on the wake width. For large wake width it is higher than unity indicating that the outlet relative flow angle is lower than the blade angle. This gives rise to specific work higher than the ideal specific work in the wake region. The slip factor variation obtained from time averaged measurements indicate that the yaw probe cannot respond to high frequency unsteady wake flows. It will respond to the jet flow and measures the average jet velocity. The slip factor obtained for the wake flow does not agree with the value of unity assumed in the jet–wake model. The new slip correlations would be helpful to evaluate the average relative flow angles in the jet and wake flows for a given jet and wake relative velocity and impeller tip speed. The slip factor obtained from measurement are lower than the values calculated from the correlations due to viscous effects. In the presence of jet–wake flow at impeller outlet, the slip factor does not remain constant across the blade pitch as suggested in the literature.

References

1. Herbert M V. Method of Performance Prediction for Centrifugal Compressors, ARC R & M 3843, 1980.
2. Came P M. The Development, Application and Experimental Evaluation of a Design Procedures for Centrifugal Compressors, *Proc. Inst. Mech. Enggs.* 1978 192, pp. 43–67.
3. Dean R C. Jr. The Fluid Dynamic Design of Advanced Centrifugal Compressors, Creare, 1976, TN–244.
4. Wiesner F J. A Review of Slip Factors for Centrifugal Impellers, *Jour. Engg Power, Trans. ASME*,, 1967, 89, pp. 558.
5. Stahler A F. The Slip Factor of a Radial Bladed Centrifugal Compressor, *Jour. Engg. Power, Trans. ASME*, 1965, 87, pp. 181.
6. Sakai J. Watanabe I. Fujie I. On the Slip Factor of Centrifugal and Mixed Flow Impellers, ASME Paper 67–WA/CT–10. 1967.
7. Whitfield A. The Slip Factor of a Centrifugal Compressor and its Variation with Flow Rate, *Proc. Inst. Mech. Enggs.* 1974, 188, pp. 415–421.
8. Stanitz J D. Some Theoretical Aerodynamic Investigations of Impellers in Radial and Mixed Flow Centrifugal Compressors. Trans. ASME. 1952, 74, pp. 473
9. Whitfield A. Wallace F J. Performance Prediction for Automotive Turbocharger Compressors, *Proc. Inst. Mech. Enggs.* 1975, 189, pp. 59-67.
10. Wislicenus G E. *Fluid Mechanics of Turbomachinery*, 1965, Dover Publications, New York.
11. Balje O E. A Flow Model for centrifugal Compressor Rotor, ASME Paper 77–GT–62, 1977.
12. Senoo Y. Maruyama S. Koizumi T. Nakase Y. Viscous Effects on Slip Factor of Centrifugal Blowers, *Jour. of Engg. Power Trans. ASME*, 1974, pp. 59–65.

Nomenclature

C	absolute velocity
d	diameter
U	peripheral speed
N_b	number of blades
m	mass flow rate
β	relative flow angle w.r.t radial
μ	slip factor
θ	flow coefficient $(4m/\pi d^2 U_2 \rho_{o1})$
ρ	density of fluid

Subscripts

b	blade
j	jet
m	meridional
o	overall
s	slip
w	wake
θ	tangential
ρ	wake relative width
λ	wake mass flow fraction